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The Effect of Heating Direction on Flow Boiling Heat Transfer of R134a in Micro-channels

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This paper presents effects of heating directions on heat transfer performance of R134a flow boiling in micro-channel heat sink. The heat sink has 30 parallel rectangular channels with cross-sectional dimensions of 500μm width 500μm depth and 30mm length. The experimental operation condition ranges of the heat flux and the mass flux were 13.48 to 82.25 W/cm^2 and 373.3 to 1244.4 kg/m^{2}s respectively. The vapor quality ranged from 0.07 to 0.93. The heat transfer coefficients of top heating and bottom heating both were up to 25 kW/m^{2}K. Two dominate transfer mechanisms of nucleate boiling and convection boiling were observed according to boiling curves. The experimental results indicated that the heat transfer coefficient of bottom heating was 13.9% higher than top heating in low heat flux, while in high heat flux, the heat transfer coefficient of bottom heating was 9.9%.higher than the top heating, because bubbles were harder to divorce the heating wall. And a modified correlation was provided to predict heat transfer of top heating.

Keywords: top heating, bottom heating, flow boiling, micro-channels, R134a, correlation

Introduction

Flow boiling has been studied in recent decades. Compared to macro-scale flow boiling, researchers increasingly put their attention on micro-scale flow boiling because of high specific surface area, high heat transfer coefficient and compact structure [1-4], these advantages are the reason of micro-scale flow boiling raising concern. Lazarek and Black [5] researched heat transfer coefficient and pressure drop for saturated boiling in a mini round tube, then proposed a new critical heat flux (CHF) correlation. Wambsganss [6] did some similar works and found high boiling number and slug flow pattern has a significant impact on nucleation mechanism.

Micro-channels heat sink can dissipate large amounts of heat from a small area [7-9], it is regarded as an efficient heat exchanger since Tuckerman and Pease [10] provided. Flow boiling in micro-channels is not similar to a single micro-channel. Flow boiling in micro-channels needs a low pressure drop in order to reduce pumping power requirement. Flow instabilities and back flow reversals in microchannels lead to undesirable characteristics such as pressure drop, flow rate oscillations, high amplitude temperature oscillations and mechanical vibration [11]. Bergles and Kandlikar [12] investigated two-phase instability in microchannels result in a lower CHF than would be got stable flow. Wang et al. [13] believed that the configuration of inlet and outlet can significantly influence the type of instability observed.

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Nomenclature

Bo	Boiling number	*x*	Vapor quality
Bd	Bond number	**Greek symbols**	
C_p	Specific heat (J kg^{-1} K^{-1})	δ	Thickness (m)
D	Channel diameter (m)	β	Enhancement factor
G	Mass flux (kg m^{-2} s^{-1})	**Subscripts**	
h	Heat transfer coefficient	*b*	Bottom heating
h_{fg}	Latent heat (J kg^{-1})	*CB*	Convective boiling
H	Micro-channels height (m)	*exp*	Experimental
k	Thermal conductivity (W m^{-1} K^{-1})	*f*	Liquid
L	Micro-channels length (m)	*g*	Gas
M	Mass flow rate (kg s^{-1})	*in*	Inlet
Q	Heat (J)	*NB*	Nucleate boiling
q	Heat flux (W cm^{-2})	*pre*	Predictive
Re	Reynolds number	*sat*	Saturated
T	Temperature (℃)	*t*	Top heating
w	Micro-channels width (m)		

Jiang et al. [14] provided three stable boiling models according to the input power level, local nucleation boiling at a lower power range and stable annual flow at higher power range was observed, but bubble flow was not observed. Peng and Peterson [15, 16] indicated cross sectional aspect ratio showed a great impact on the flow friction and convective heat transfer both in laminar and turbulent flow. Matthew Law [17] presented a new oblique-finned microchannels, nucleate boiling was the dominate heat transfer mechanism at low heat fluxes when a transition to convection boiling-dominant at medium heat fluxes, and a full convection boiling mechanism was observed before CHF. This novel oblique fine could make the flow boiling process stable. Yang et al. [18] considered inlet restrictors was an important factor to enhance the average heat transfer coefficient in micro-channels, it could be enhanced by up to 326% ignored inlet restrictors, while the average heat transfer coefficient only could be enhanced by up to 248%. T Harirchian [19] developed a comprehensive flow map, consisted four different regions – bubbly, slug, confined annular, and alternating churn/annular/wispy-annular flow regions, and it also proved flow regime-based modeling could predict heat transfer and pressure drop reliably. Many researchers used experimental and numerical methods to research the heat transfer and pressure drop characteristics of nanofluids in micro-channels [20-24].

Many researchers provided several correlations about flow boiling in micro-channels. Warrier et al [25] presented subcooled and saturated nucleate boiling in small rectangular channels using FC-84 as test fluid, and developed a correlation dependent on *Bo* number and vapor quality *x* to predict saturated nucleate boiling. Agostini and Bontemps [26] performed ascendant forced flow boiling in a 2.01mm channels with R134a, they correlated the heat transfer coefficient with *G*, *q* and *x*. Li and Wu [27] did a wide range of operational conditions, they use water, refrigerants, FC-77, ethanol and propane as fluid work, collected more than 3700 data point. They considered *Bo* number, *Bond* number and liquid Reynolds number had an impact on heat transfer coefficient.

Generalizing from the above works, many researchers have done many extensive researches in micro-channels. However, almost all researchers did not consider the influence of heating directions on flow boiling heat transfer in microchannels. Therefore, this paper used top heating and bottom heating to study effect of heating direction on flow boiling heat transfer of R134a in micro-channels.

Experimental apparatus and method

Experimental apparatus

The experimental setup is shown in Fig. 1. The apparatus is a close loop system for stable and smooth flow as well as convenient adjustment. The system consists of a fluid reservoir, pump, filter, two mass flow-meters, two visual liquid mirrors, expansion valve, plate heat exchanger, chiller, test section and data acquisition system. Working fluid R134a was driven by a pump and flowed through the filter, entered the test section heated by the heating system, and then was cooled down by the plate heat exchanger.

The test section was a brass heat sink with 30 channels and the dimension of each channel was 500 μm width 500 μm depth, and 30 mm length. The back of test section had a rectangular groove(30 mm × 26 mm) in order

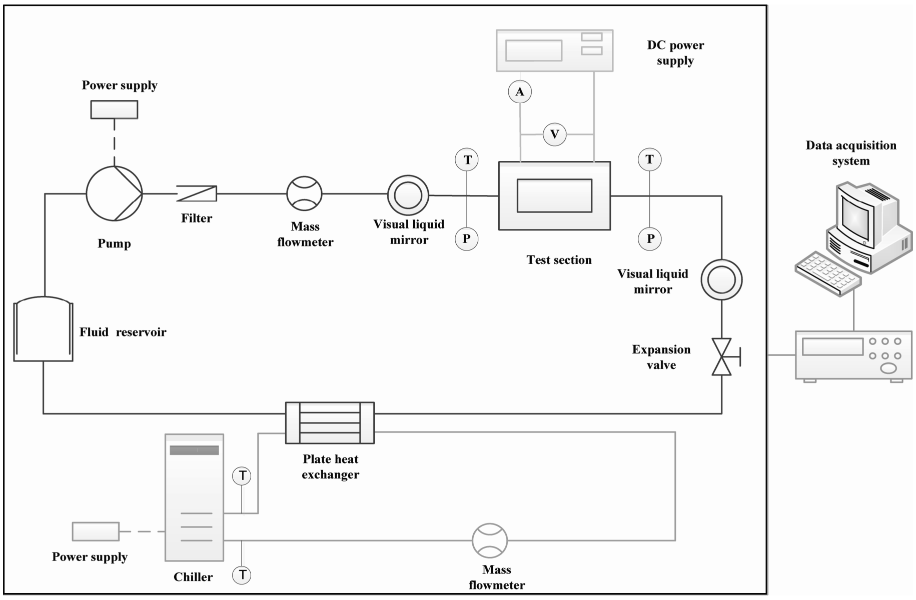


Fig. 1 Experimental setup

to reduce thermal resistance, and the thickness between micro-channels and heater is 1mm. The back had three shallow channels, there were three thermocouples in every channel, using silver paint to fix in order to contact with the wall well and reduce the thermal resistance. The inlet and outlet of test section had a thermocouple and pressure sensor to measure the temperature and pressure of inlet and outlet.

Data processing

The flow boiling heat transfer coefficient h is calculated from:

$$h=\frac{Q}{A\left(T_w-T_{sat}\right)} \tag{1}$$

Q is the heat of micro-channels, T_{sat} is the liquid saturation temperature, and T_w is the temperature of micro-channels, however, thermocouples measure the temperature of heat sink wall T_s, the relation between T_w and T_s is :

$$T_w=T_s-\frac{q\delta}{k} \tag{2}$$

δ is the thickness from heat sink heated surface to the surface of micro-channels bottom. q is heat flux and k is the thermal conductivity of brass. A is the total heated area in the micro-channels:

$$A=N(w+2H)L \tag{3}$$

Where N is the number of micro-channels in heat sink, w, H, L are the width, depth, and length of micro-channels respectively. In this experiment, thermodynamic vapor quality is used to describe the vapor quality of flow boiling in micro-channels which is defined as:

$$x=\frac{Q-Mc_p\left(T_{sat}-T_{in}\right)}{Mh_{fg}} \tag{4}$$

Where h_{fg} is the latent heat of the working fluid, M is the mass rate, and C_p the specific heat capacity at constant pressure, T_{sat} is the saturated temperature in the test section and T_{in} is the temperature of inlet.

It also need an analysis of experimental data, the uncertainly range of experimental instruments and data acquisition as shown in Table 1.

Table 1 Uncertainty of experimental parameters

Parameter	Uncertainty	Parameter	Uncertainty
Channel diameter	±0.002mm	Absolute pressure	±0.05%
Temperature	±0.3℃	Mass flow rate	±2%
Heat flux	±2%	Mass flux	±2%
Heat transfer coefficient	<±14%		

Results and analysis

In this study, a wide range of experimental conditions was used to research. The range of mass flux was from 373.3 to 1244.4 kg/m^{2}s and heat flux was from 13.48 to 82.25 W/cm^2.

Effect on micro-channels temperature

Fig. 2 shows the relationship of wall temperature under two kinds of heating directions variation with heat flux at different mass flux. Under high mass flux as the heat flux increase, the rise of wall temperature is not evident, which illustrate that the effect of heat transfer is

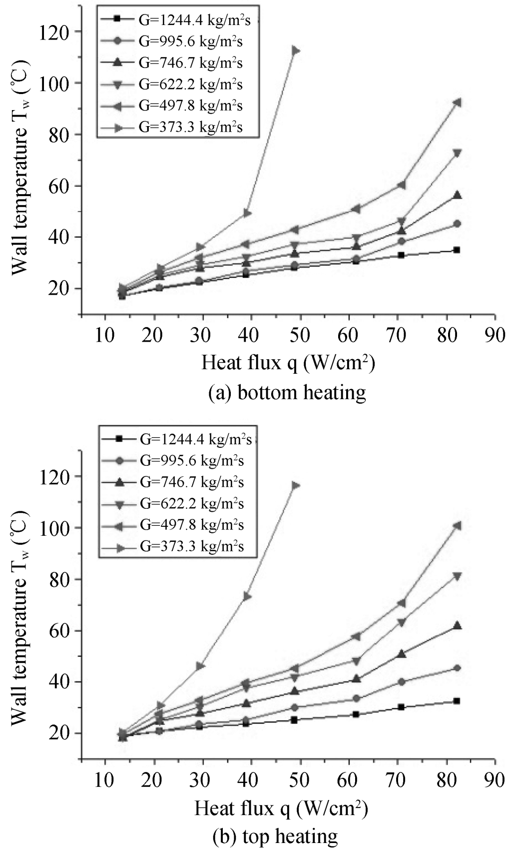


Fig. 2 Micro-channels wall temperatures versus heat flux

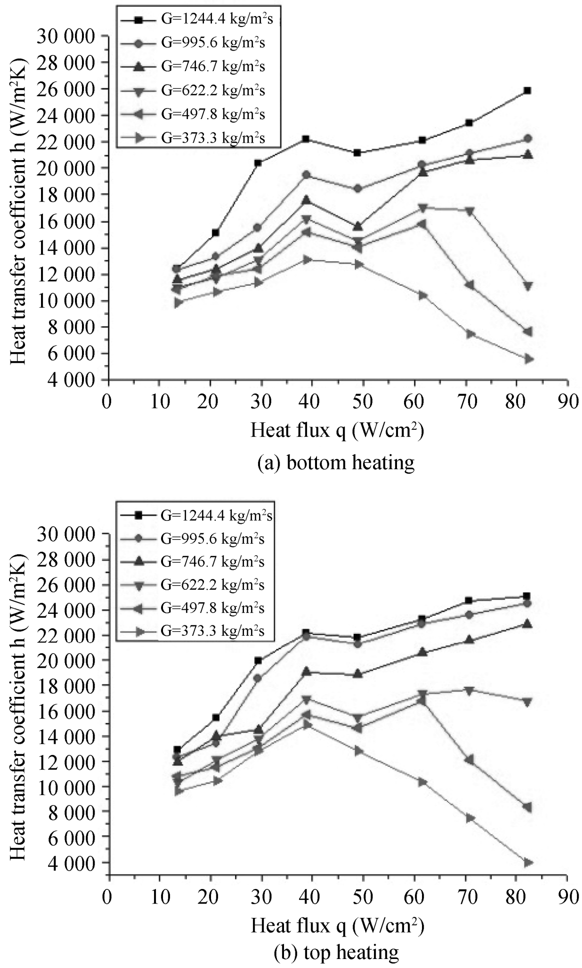


Fig. 3 Heat transfer coefficient as a function of heat flux for different mass fluxes

excellent; however, as the mass flux decrease, the increase of wall temperature become more and more remarkable with the increase of heat flux. Under high heat flux (61.54 W/cm^2–82.25 W/cm^2), there is a sudden rise on the temperature curve, which indicate that partial dry out may occur on the micro-channel surface. Under the low mass flux (373.3 kg/m^{2}s), wall temperature reach above 110℃ when the heat flux is 48.9W/cm^2, partial dry out phenomenon emerged. If we continue to heat the test section, the wall temperature will increase sharply. The operating temperature of electronic device like IGBT is supposed to be kept under 120℃, so that we must prevent the situation that the element gets a high heat flux while the mass flux of refrigeration is low, and partial dry out should be avoided too. Furthermore the trends of wall temperature under different heating direction are almost the same.

Effect on heat flux

The relationship between heat transfer coefficient and heat flux for different mass fluxes with two heating methods is shown as Fig. 3. The two curves have a similar tendency. The analysis of flow boiling mechanism, which is given by T. Chen and S.V.Garimella [27], indicates that heat boiling coefficient tends to increase with heat flux when the latter is between 13.48 W/cm^2 and 38.89W/cm^2, in which case the forced convection heat transfer is dominant and the heat transfer is enhanced. When the heat flux is between 38.89 W/cm^2 and 48.89 W/cm^2, the heat transfer coefficient decreases for the local heat transfer deterioration caused by intermittent flow, which appears when the nucleate boiling turn to forced convection heat transfer. In this process, the formation of plug flow and slug flow is caused by bubble growth and coalescence. When the heat flux is between 48.89 W/cm^2 and 82.25 W/cm^2, the heat transfer coefficient increases with the heat flux for the forced-convection boiling is playing a primary role in the boiling heat transfer mechanism. The heat transfer is enhanced. Things are the opposite for the medium and low mass flux for the dry out phenomenon, which leads to the heat transfer deterioration.

Effect on vapor quality

Fig. 4 shows the correlation between heat transfer coefficient and vapor quality at different mass flux of both heating direction. At the low heat flux, nucleate boiling is

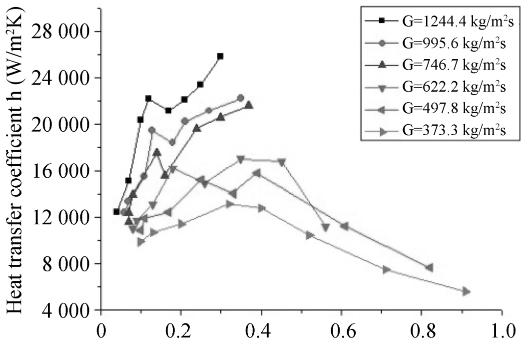
not sufficient and the vapor quality increased slowly. With the increase of the heat flux, nucleate boiling is more intense, the vapor quality increased quickly and the heat transfer coefficient increased to the local maximum. This variation is the same to the correlation in Fig.4. Then the heat flux continues to increase, the flow regime will change and the heat transfer coefficient declined slightly with the increase of the vapor quality. Continuing heating and the forced convection boiling will be dominate, at the same time, the annular flow occurs in micro-channels and the vapor quality increases steadily. In the low mass flux and high heat flux the liquid film will be destroyed. Besides, the vapor quality increases quickly and the heat transfer coefficient decreased with the partial dry out occurs.

Effect on different heating direction

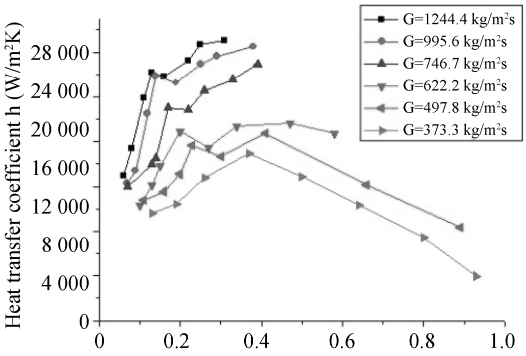
The heat transfer performances of different heating directions are compared in this study as shown in Fig. 5. It is observed that the heat transfer performance of bottom heating is better than that of top heating. During low heat flux q=20.59 W/cm^2, the heat transfer performances of top heating and bottom heating are basically stable. The heat transfer coefficient of bottom heating is higher by 13.9% as opposed to that of top heating. Nevertheless, both heat transfer coefficients are relatively low in high heat flux q=59.16 W/cm^2 and low mass flux. The heat transfer coefficients increase as the mass flux increase and both heat transfer performances are enhanced. However, with the increase of mass flux, both the heat transfer coefficients increase at first, and keep unchanged basically then. But the heat transfer coefficient of bottom heating is 9.9% higher than that of top heating. It is found that the heat transfer coefficient of bottom heating is always higher as compared with that of top heating regardless of the high and low heat flux. The bubbles that emerge, grow and coalesce during flow boiling slip and depart from heated wall, and flow on the top of channels due to the effect of buoyancy, Thus the contact area with liquid refrigerant in bottom heating are larger than that in top heating. Furthermore, the generated bubbles are difficult to depart from upper wall surface of channels because of the influence of buoyancy. The phenomenon become not obvious with the increase of heat flux, because annual flow usually occurs in higher heat flux, and the number of bubbles decrease. The effect of heating direction was evident in low heat flux. The partial dry out gives rise to decrease the heat transfer coefficient in high heat flux and low mass flux.

This paper provides an enhancement factor β to represent the enhancement of bottom heating, it can be defined as follows:

$$\beta=\frac{h_b-h_t}{h_t} \tag{5}$$



(a) bottom heating



(b) top heating

Fig. 4 Heat transfer coefficient as function of vapor quality for different mass fluxes

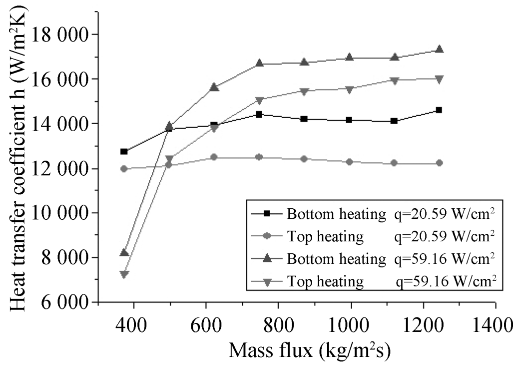


Fig. 5 Heat transfer coefficient as a function of mass flux for different heat fluxes

Fig.6 shows the degree of enhancement which bottom heating mode compared to top heating mode under different mass flux. Enhancement factor β increases with the increase of mass flux, and then stabilized after mass flux reaches a certain value. It indicates that the bubbles attached on the heated surface are difficult to depart and be taken away by the working fluid under lower mass flux. It shows an adverse mechanism compared with lower mass flux when the mass flux is higher. In this case the mass flux is not the main factor affecting heat transfer

coefficient and enhancement factor β becomes stable. However, enhancement factor β decreases with the increase of mass flux at higher heat flux. Flow pattern within the channel is annular flow and the number of bubbles reduced when the heat flux is higher. Therefore, enhancement factor is small and the effect of heating direction on heat transfer coefficient is unobvious under the condition of lower mass flux. Both high-heat flux and low-mass flux is beneficial to bubble growing and aggregation, which will affect the efficiency of heat exchange and result in enhancement factor β increases.

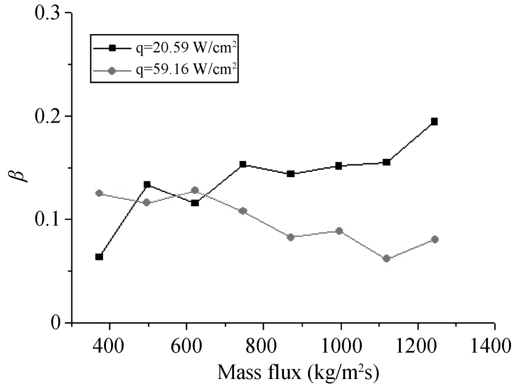


Fig. 6 Enhancement factor β as a function of mass flux for different heat fluxes

Correlation predicted and modified

Some researchers have proposed a number of correlation formulas about flow boiling in micro-channels, these correlation formulas are validated by the data acquired from the experiment in this study. The comparisons of experimental data collected from bottom heating and top heating with predictions of some correlation formulas are shown in Fig. 7 and Fig. 8 respectively, and the predictive accuracy of a correlation is measured by the mean absolute error, it is defined as follows:

$$MAE = \frac{1}{N}\sum\left[\frac{\left|h_{pred} - h_{\exp}\right|}{h_{\exp}} \times 100\%\right] \tag{6}$$

Under the condition of bottom heating, the prediction of the correlation formula proposed by Lazarek and Black [5] is smaller compared to experimental data, the value of MAE is 48.9%. The predictor based on the correlation formula presented by Li and Wu [2] is also smaller than the data acquired from experiment and the MAE is 42.9%. While the prediction of the correlation formula proposed by Warrier [25] is much smaller as opposed to experimental data and the MAE become 91.4%. It is observed that the predictions of the above-mentioned correlation formulas are generally smaller than the experimental data, the relative deviations exceed 30%. Nevertheless, when the experimental data is com-

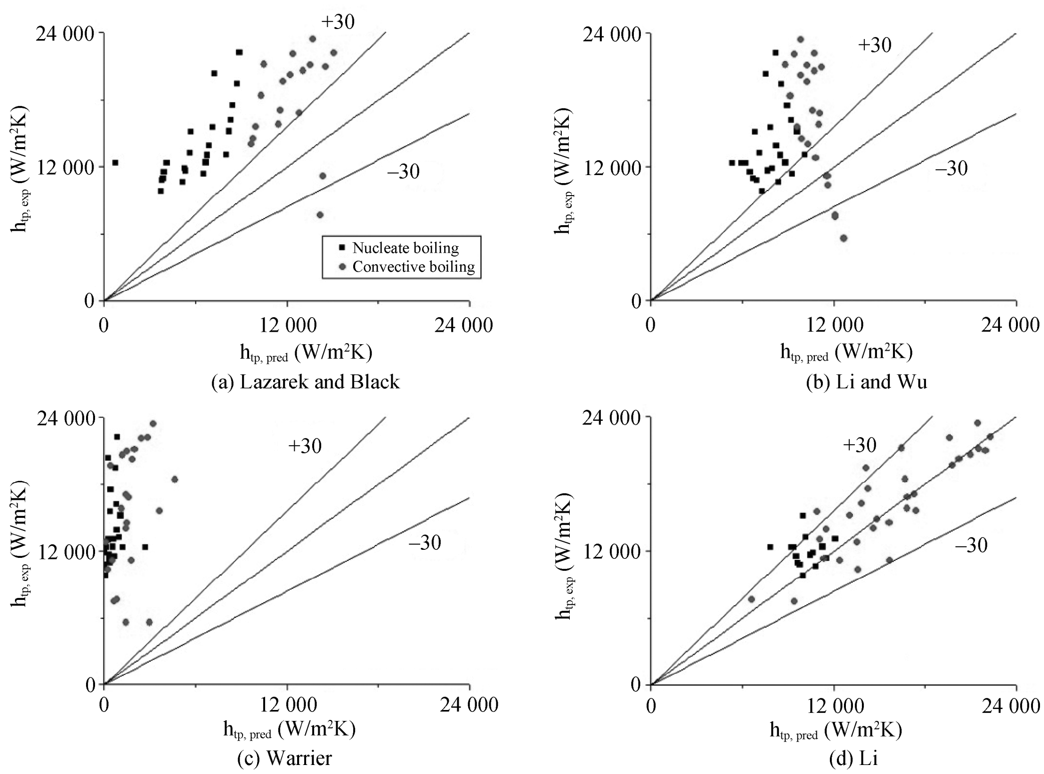


Fig. 7 Comparison between trends in the measurements and those from the proposed correlation in bottom heating

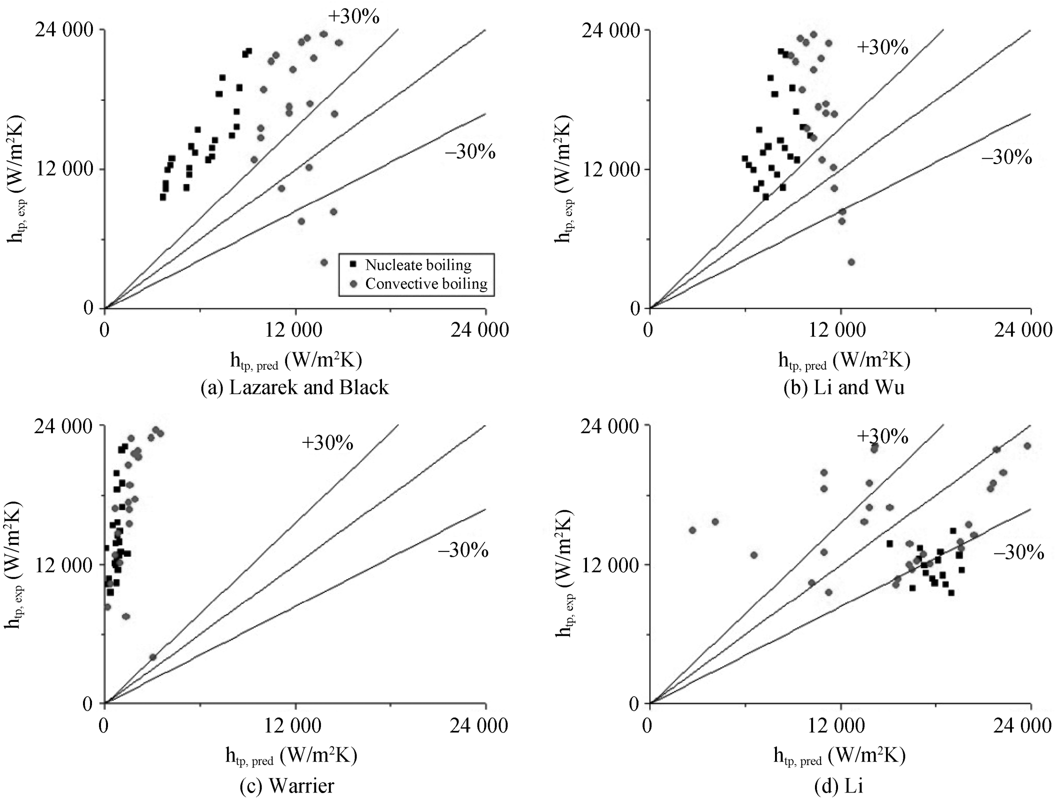


Fig. 8 Comparison between trends in the measurements and those from the proposed correlation in top heating

pared with the predictor of the correlation presented by Li [28], the predictor match well with experimental data and the value of MAE is 12.6%. It is found that the correlation proposed by Li can estimate accurately the heat transfer coefficient of bottom heating. In the top heating, the predictor of the correlation proposed by Lazarek and Black is smaller than experimental data and the value of MAE is 51.8%. The prediction based on the correlation presented by Li and Wu is also smaller as compare to experimental data, the MAE become 47.1%. However, the prediction of the correlation proposed by Warrier is much smaller than experimental data and the value of MAE increase up to 90.9%. It is apparent that predictions of the aforementioned correlation formulas don't agree well with experimental data and the error exceed 30%. The predictor of the correlation proposed by Li is larger compared to experimental data and the value of MAE becomes 39.2%. It is evident that the correlation presented by Li is not able to estimate accurately the heat transfer coefficient of top heating, the correlation must be modified.

Bubbles are difficult to depart from the heated surface duo to the effect of buoyancy under top heating mode, which will suppress the degree of boiling at a certain extent. Therefore, the correlation can be modified through correcting the number of *Bo*. The corrected correlations as showed:

$$h_{NB} = 189 Bo^{0.145} Bd^{0.4}\, \mathrm{Re}_f^{0.12} \left(\frac{k_f}{D}\right) \tag{7}$$

$$h_{CB} = 277.3\, \mathrm{Re}_f^{0.94}\, Bo^{0.993} (1-x)^{0.47} \left(\frac{k_f}{D}\right) \tag{8}$$

Experimental data and the result that predicted according to the corrected correlation are compared in Fig. 9, which shown excellent consistency between them. The MAE is 16.6% and the error within 30% for both nucleate boiling and forced convection boiling.

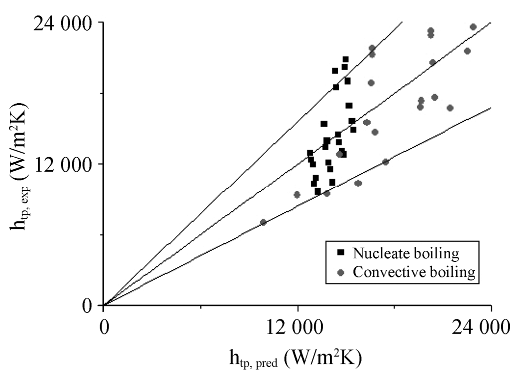


Fig. 9 Comparison of predictions of suggested correlations with experimental data

Conclusions

This paper researches the effect of heating direction on flow boiling heat transfer of R134a in micro-channels. Experiment were conducted with the mass flux from 373.3 to 1244.4 kg/m^{2}s, and heat flux ranging from 13.48 to 82.25 W/cm^2. Major conclusions from the current study are given as follows:

1. The wall temperature of micro-channels did not rise obviously in a higher mass velocity, the effect of heat transfer is good, but in a low mass velocity, the wall temperature rise evidently, even it happened dry out in micro-channels in a high heat flux .

2. Both these two heating directions had emerged two dominate transfer mechanisms, nucleate boiling and convection boiling were observed according to boiling curves. When nucleate boiling transferred to convection boiling, flow pattern changed, resulting in the deterioration of local heat transfer, the flow boiling heat transfer coefficient decreased.

3. Mass velocity, heat flux and vapor quality had an important impact on heat transfer in micro-channels. When mass flux increasing, the heat transfer coefficient was increasing too. Heat flux and vapor quality had remarkable influence on flow pattern change in micro-channels.

4. Comparing two heating direction, the heat transfer coefficient of bottom heating was higher than the heat transfer coefficient of top heating. The heat transfer coefficient of bottom heating was 13.9% higher than top heating in low heat flux, while in high heat flux, the heat transfer coefficient of bottom heating was 9.9%.higher than the top heating. The bubbles moved upward easily because of buoyancy, bubbles were harder to divorce the heating wall in top heating, but in a high heat flux, annular flow was dominant in micro-channels, the number of bubbles was decreased, the effect of heating direction was obvious in low heat flux.

5. A modified correlation was proposed to predict heat transfer of top heating, and the MAE of modified correlation was 16.6%.

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